

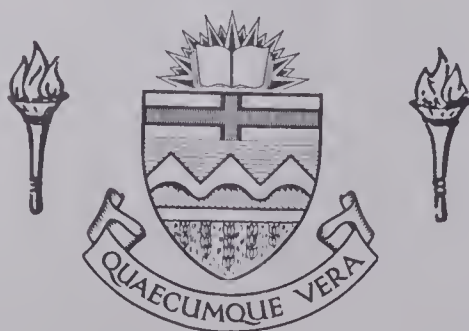
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DESIGN OF A LOW-SPEED WIND TUNNEL
WITH BOUNDARY LAYER CONTROL DIFFUSERS

by



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A THESIS

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The undersigned certify that they have read, and recommend to the Faculty of Graduate Studies for acceptance, a thesis entitled "Design of a Low-Speed Wind Tunnel with Boundary Layer Control Diffusers" submitted by Lawrence A. Schienbein in partial fulfilment of the requirements for the degree of Master of Science.

ABSTRACT

The study proceeds from a discussion of low-speed wind tunnel design to the application of wide angle boundary layer control diffusers in the design of a low-speed closed circuit wind tunnel. The pressure loss in the wind tunnel using boundary layer control diffusers is found to be approximately half the pressure loss in an equivalent wind tunnel of more conventional design, while the space required by each tunnel is found to be very nearly equal.

ACKNOWLEDGEMENTS

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LIST OF SYMBOLS

A	area of the tunnel component
A_f	cross sectional area at the fan
A_{ff}	flow area at the fan
A_m	largest median area of a model
A_{nf}	largest cross sectional area of the fan nacelle
A_{ts}	area of the test section
C	specific heat of air
C_x	model drag coefficient including supports
C_{xf}	drag coefficient of the nacelle
D_f	fan diameter
D_h	hydraulic diameter
K	total hydraulic resistance coefficient of the component
K_d	resistance coefficient of the fan duct
K_n	resistance coefficient of the nacelle
K_s	resistance coefficient of the screen
K_{ts}	total hydraulic resistance referred to the test section
L	axial length of the section
P	rated motor power output
V	maximum test section speed
a, b	dimensions of a rectangular section
dT	temperature rise of the stream per circuit
n	contraction ratio

LIST OF SYMBOLS (continued)

η	fan efficiency
ρ	density of air
λ	coefficient of frictional resistance

CHAPTER I

INTRODUCTION

1.1 Discussion

The purpose of this design study is to present a unified approach to the design of a low-speed wind tunnel. The study will emphasize the problems facing the design engineer in the attempt to apply existing information to the design of low-speed wind tunnels.

The first step in the design is to determine the purpose or purposes of the device. To fulfill the purposes, certain requirements are made of the wind tunnel. At the same time the designer must consider factors such as the efficiency, the available space, and the cost.

For this study, the space available for the wind tunnel is given as an area 75 feet long, 24 feet wide, and 27 feet high. An attempt will be made to achieve an energy ratio of five while keeping the installed motor power in the neighborhood of 150 H.P. (The energy ratio is here defined as the ratio of the wind power in the test section to the installed motor power). In addition, it is expected that the wind tunnel will be used for a wide variety of tests.

The ultimate goal of the study is the determination of the energy ratio of the wind tunnel and the draft layout of the tunnel. Because the tunnel layout is primarily determined by the testing requirements, the study will proceed in general from a discussion of testing

requirements to the determination of the tunnel dimensions and finally to the calculation of the energy ratio.

1.2 Types of Wind Tunnels

Two main configurations are available for wind tunnels: open return and closed return. The choice of configuration is made on the basis of available space, overall efficiency, and the requirements of flow uniformity at the inlet to the settling chamber.

The open return wind tunnel, shown in Figure 1.1, consists of a settling chamber, an inlet contraction, a diffuser, and a fan at either the inlet or the outlet. The closed return wind tunnel, shown in Figure 1.2, consists of a settling chamber, a contraction, a test section, and one or more diffusers which make up the return passage linking the test section exit with the settling chamber inlet.

The important advantage of the open return tunnel is that contaminated or greatly disturbed air (strong model wakes) cannot be recirculated and reappear in the test section. In addition, no significant problems are connected with the addition or the extraction of air for the purpose of boundary layer control on a model or on the tunnel walls.

The primary disadvantages of the open return tunnel are the noise, the possible disturbing cross flow in the test section through leaks in the wall (the test section static pressure is less than the ambient pressure), and the inability to operate with an open test section without the addition of a sealed chamber surrounding the test

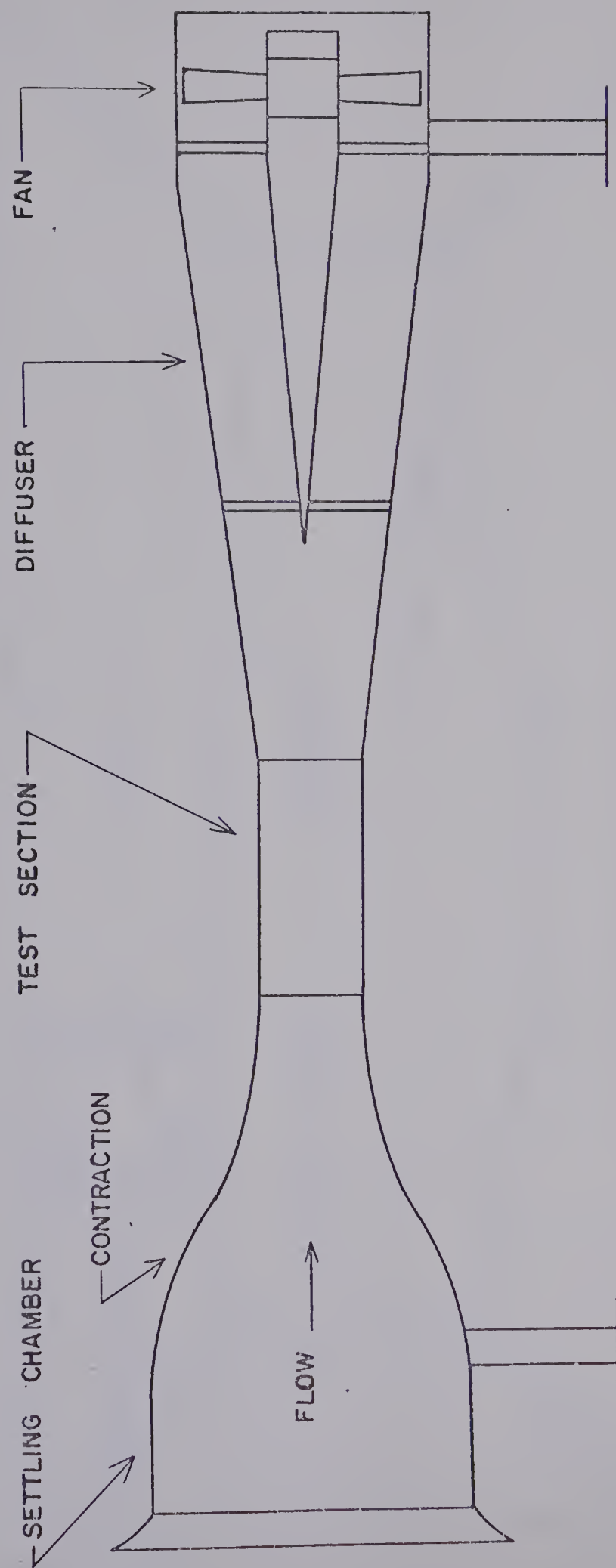


FIGURE 1.1 TYPICAL OPEN RETURN WIND TUNNEL

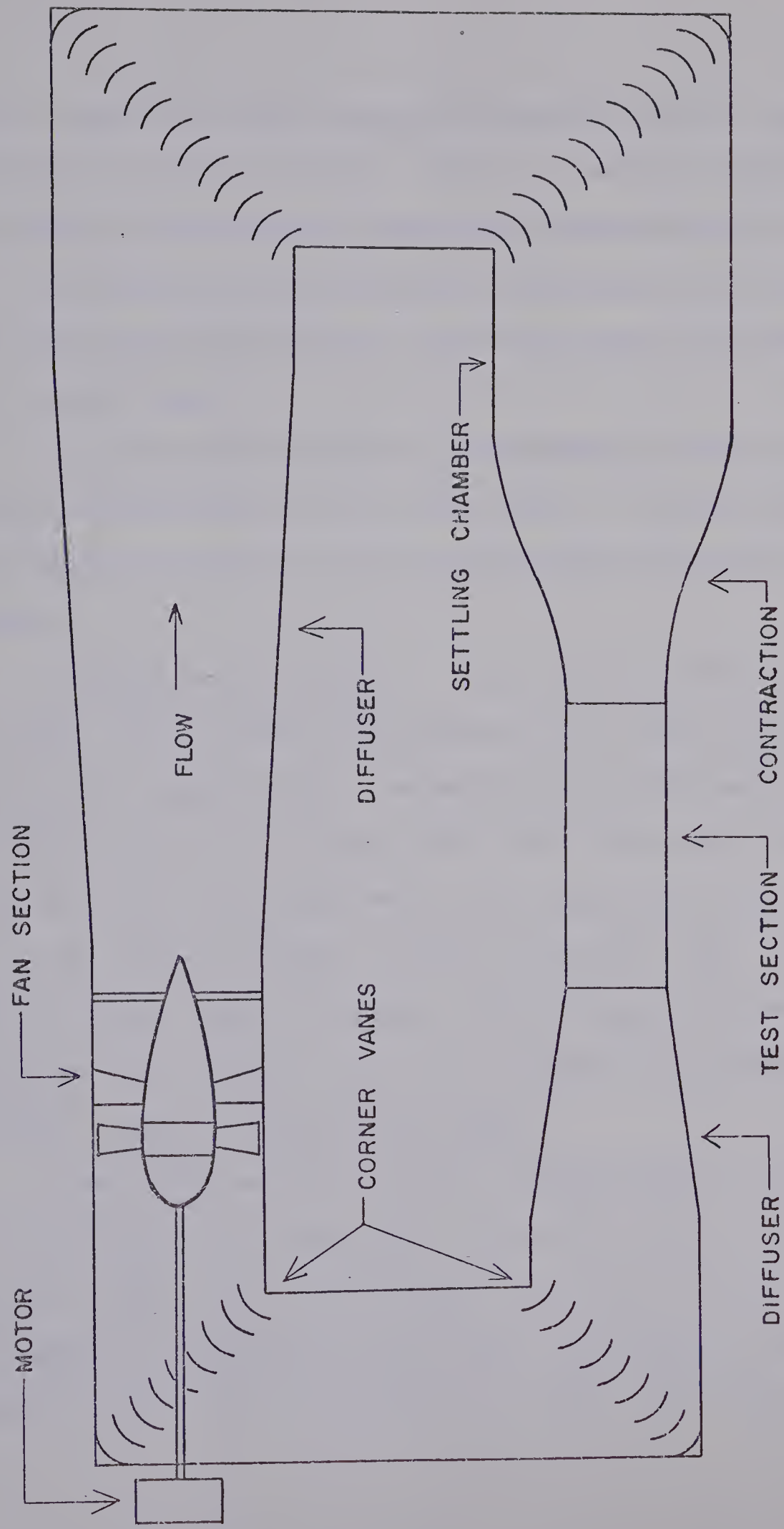


FIGURE 1.2 TYPICAL CLOSED RETURN WIND TUNNEL

section. However, the cross flow can be controlled, and few types of tests require an open test section. Noise is a testing problem if the noise results in the creation of turbulence in the test section.

The closed return tunnel can be more easily modified to operate with an open test section. The return circuit should help to reduce the noise level.

The closed return tunnel is the best choice in the attempt to fulfill the space and efficiency requirements. A closed return tunnel with the same energy ratio as an open return tunnel will occupy less space.

Consider two wind tunnels: an open return tunnel and a closed return tunnel with the same inlet contraction ratio, test section area, and maximum test section speed. The energy ratio of the open return tunnel will in general be less than that of the closed return tunnel. In order to increase the energy ratio of the open return tunnel, the length of the settling chamber, the inlet contraction ratio, and the diffuser area ratio must be increased. These changes are necessary to decrease the outlet pressure loss and the pressure loss associated with flow control screens in the settling chamber.

Open return tunnels are usually located indoors so that the ambient conditions can be controlled. The return circuit is then the free space of the room. The flow quality in the test section is affected by the height of the tunnel above the floor, the distance of the air intake from the wall, and any obstacles in the room. Gorlin and Slezinger [1]

suggest that the width of the room should be five or six times the equivalent intake diameter to ensure satisfactory flow in the test section.

Using the preceding figure as a guide, and assuming that the open return tunnel may be up to twice as long as the closed return tunnel, we find that the open return tunnel introduced previously, having as high an energy ratio as the closed return tunnel, would require a minimum floor area of two to three times the area required by the closed return tunnel.

The open return type of wind tunnel would be the best choice if unlimited space were available.

1.3 Open and Closed Test Sections

In the preceding section it was indirectly suggested that the wind tunnel should operate with a closed test section. The closed test section is clearly the best choice.

The pressure loss associated with the closed test section is much less than that associated with the open test section. With the open test section, stagnant air is entrained by turbulent mixing at the jet perimeter. The momentum lost by the main stream is equivalent to a substantial increase in the effective surface friction coefficient. At the same time the efficiency of the first diffuser may be decreased because of the relatively thick boundary layer at the diffuser inlet. Pope and Harper [2] state that the power required to operate a wind tunnel with an open test section may easily exceed three times the power

required by the same tunnel operating at the same speed with a closed test section.

The wall corrections are not really simplified with an open test section. One solid boundary is simulated, since the wind tunnel force balance and other measuring devices must be shielded. This tends to confuse the boundary corrections and raises the question as to whether the test section is really open.

Aerofoil and half models are more easily mounted in the closed test section because they can be attached to the walls. However, models are more accessible in an open test section and this is the only real advantage of the open test section.

CHAPTER II

TESTING

2.1 Proposed Uses

Wind tunnel testing can be divided into two main categories: aerofoil testing and model testing.

Preliminary discussions of the proposed uses of the wind tunnel resulted in the development of the following list:

A. Aerofoil Testing

- (i) development of aerofoil sections
- (ii) development of high-lift flaps and boundary layer control sections

B. Model Testing

- (i) aircraft models
- (ii) automobile and boat models
- (iii) air cushion and V/STOL aircraft
- (iv) industrial aerodynamics

2.2 Testing Requirements

The basic requirement for all wind tunnel testing is that the velocity in the test section be uniform and rectilinear, providing a large "core" in which the model can be placed. At the same time the static pressure should be as nearly constant as possible throughout the

test section.

In addition to the aforementioned, there are three major requirements for each type of test. These requirements are:

1. A specific minimum Reynolds number based on some dimension should be achieved.
2. A particular model size to test section size ratio should not be exceeded. A particular test section shape may also be required.
3. A definite maximum turbulence level in the test section should not be exceeded.

2.2.1 Reynolds Number

Bradshaw and Pankhurst [3] suggest that, as a general rule, accurate quantitative work on three-dimensional aerofoils and other bodies is not easy in wind tunnels where the test section Reynolds number based on the equivalent diameter is less than 2×10^6 . The underlying assumption is that in full scale flight laminar boundary layer separations do not occur; most of the boundary layer is assumed to be turbulent. The preceding rule then follows directly from Preston's value for the minimum Reynolds number at which a fully developed turbulent boundary layer can exist. Preston's value is a Reynolds number of 320 based on the boundary layer momentum thickness.

This Reynolds number criterion is mainly required for aerofoil and aircraft model testing where a specific minimum Reynolds number is necessary so that extrapolation may be carried out to higher Reynolds numbers. In the normal low-speed flight range, the lift and drag co-

efficients are practically independent of the Reynolds number.

Testing at the full scale Reynolds number is not important and is, indeed, undesirable in other types of tests. For example, in testing models of buildings or other bluff bodies, the flow will separate at sharp corners almost regardless of the Reynolds number, so that the flow pattern can be accurately represented with any convenient wind speed. Some Reynolds number dependent phenomena occur at low Reynolds number and can be easily simulated. The Von Karman vortex street is an example.

2.2.2 Model Size

With a closed test section, up to nine corrections may have to be made to the test data because of the existence of the lateral boundaries (see reference [2]). Some of the corrections are reduced in magnitude by a decrease in the ratio of model cross sectional area to the test section cross sectional area. In addition, some corrections, such as the corrections due to the effect of the downwash of a high-lift wing section, are diminished by using a deeper test section.

2.2.3 Turbulence Level

Large scale air velocity fluctuations in the atmosphere are very pronounced (gusts); however, small scale fluctuations (turbulence) have been shown to be very small in magnitude. A high test section turbulence level results in turbulent boundary layers forming on models at lower Reynolds numbers than at full scale atmospheric conditions.

Therefore, for accurate simulation of full scale conditions, a considerable effort must be made to achieve a very low turbulence level in the test section.

In some cases the attainable Reynolds number may be low and turbulence may then be beneficial in that it increases the effective Reynolds number.

2.3 Aerofoil Testing

Aerofoil testing is two-dimensional in nature. Usually a constant chord section is made to span the shorter dimension of a rectangular test section or is made to span the distance between two flat inserts in a test section of unsatisfactory shape.

The primary requirement of aerofoil testing is the Reynolds number requirement. A Reynolds number of 2.5×10^6 (based on the model chord) need not be exceeded for most purposes. Some compromise must be made between test section speed and model size in order to achieve this Reynolds number. It is not advisable to exceed a flow velocity of 250 to 300 feet per second in low-speed wind tunnels because beyond this range compressibility effects and wind loading on tunnel components become a problem.

A simple calculation based on a Reynolds number of 2.5×10^6 , a flow velocity of 250 feet per second, density equal to 0.0022 slugs per cubic foot, and a coefficient of viscosity of 3.68×10^{-7} slugs per foot second yields an aerofoil chord length of 1.67 feet, which is a convenient size.

The size of test section required can now be determined by considering the wall corrections. Many of the wall correction factors for two-dimensional testing (see reference [2]) include the factor c/h , where c is the model chord length and h is the test section dimension perpendicular to the aerofoil section. The corrections can be diminished to the point where they can often be neglected by setting c/h less than about $1/5$. The solid blockage correction is, of course, greatly reduced by increasing the ratio of the test section area to the equivalent blockage area of the model.

The same restrictions applying to general aerofoil testing also apply to boundary layer control models, because the lift-constraint corrections for boundary layer control sections can be determined by the same methods used to determine the corrections for general aerofoil sections. (see reference [2])

The recommended side ratio of the test section for aerofoil testing is in the neighborhood of 2.5:1. (The NACA wind tunnel designed for aerofoil testing has a side ratio of 2.5:1). Given this side ratio, a depth to chord ratio of 5:1, and a chord of 1.67 feet, the width of the tunnel should be 3.34 feet and the depth should be 8.35 feet.

High lift flap testing raises the problem of the downwash interacting with the boundaries. The impingement of the jet on the tunnel floor and a possible change in the assumed vorticity distribution rule out the conventional correction treatment. Williams and Butler [4]

suggest that a partially vented working section with slotted or perforated walls offers a considerable reduction in lift coefficient and blockage corrections.

2.4 Model Testing

If the model to be tested has a plane of symmetry, it is usually advantageous to use a half model, which effectively doubles the equivalent working section size. This allows more model detail, higher Reynolds numbers, and more room for instrumentation.

2.4.1 Aircraft Models

Aircraft model testing requires a Reynolds number of about 2.5×10^6 based on the mean chord length. The model span should be kept below 0.8 times the test section span to avoid excessive wall corrections. A test section width to depth ratio of 1.5:1 is recommended. (see references [1] and [2])

2.4.2 Automobile Models

Tests performed on automobile scale models determine the basic aerodynamic forces. Full-size automobiles must be used when testing components such as the radiator. The wind speed in the test section should be variable from zero up and the suggested side ratio for the test section is 2:1. (see reference [5])

2.4.3 Boat Models

The test section length should be from two to four equivalent

diameters to accommodate the long hulls. Proper venting of the test section will reduce the adverse pressure gradient and the corrections for horizontal buoyancy.

2.4.4 V/STOL Models

V/STOL models should have a span of at least six feet so that many details can be displayed and so that sufficient internal space is available for instrumentation. This size of model requires a test section at least 15 feet square to minimize the high wall interference effects.

An open return tunnel is desirable for V/STOL testing because of the possibility of the wake or exhaust gases reappearing in the test section of a closed return tunnel. V/STOL models should be tested in conventional low-speed wind tunnels for cruise conditions only with transition and hovering flight tests conducted in much larger test sections. (see references [6] and [7])

2.4.5 Industrial Aerodynamics

The problems of industrial aerodynamics are the relations of the natural wind to large industrial structures. The objects to be tested should be large enough so that all the relevant detail is represented. In practice this implies a long working section of rectangular form. Although high speed is not necessary, the speed should be variable from zero up. The side ratio of the test section should be about 2:1.

Provision should be made in the test section for apparatus (rods and screens) to introduce wind shear and turbulence; the floor must be artificially roughened to produce a turbulent boundary layer at low speeds. (see reference [8])

2.5 The Test Section

The cross sectional area of the test section is in part determined by the allowable motor power. The maximum allowable test section area, given a maximum installed motor power of 150 H.P. and assuming an energy ratio of 5:1 and a maximum test section air speed of 250 feet per second, is 24 square feet. The area of the aerofoil test section determined in section 2.3 is 27.9 square feet.

A test section side ratio of 2.5:1 is too large for general model testing as the boundary correction problems increase with increasing side ratio. A test section side ratio of 1.4:1 is the most popular ratio for general purpose wind tunnels. However, since the suggested side ratio for automobile and industrial model testing is 2:1 and the recommended side ratio for aerofoil testing is 2.5:1, a compromise choice for the side ratio is a ratio of 2:1.

For our tunnel the test section dimensions will be 3.25 feet by 6.50 feet. These dimensions give a cross sectional area of less than 24 square feet. The area has been set at less than 24 square feet because the assumed energy ratio may be an optimistic estimate. The dimensions give a depth to chord ratio of 3.9:1 for an aerofoil chord of 1.67 feet.

The test section should be long enough so that strong wakes produced by models have time to dissipate before the diffuser inlet. A long test section of perhaps four equivalent diameters in length is valuable when long models such as boat models are to be tested. However, since the bulk of the testing will likely not require an extremely long test section and since the test section does contribute to the pressure loss, a test section length of about two equivalent diameters (10 feet) is sufficient.

2.5.1 Test Section Shape

Three test section shapes are commonly used: rectangular, elliptic, and octagonal. The best choice is the rectangular cross section.

The elliptic test section gives the greatest span for a given test section area, while the octagonal section eases the problem of transition from the test section to the circular fan section. However, model mounting and model viewing are difficult with both of these sections.

The rectangular section allows models to be mounted on the walls, if necessary. In addition, the walls can be made removable so that a model can be mounted on a wall and inserted into the test section immediately after a different model has been removed. The test section walls can be made of a heavy transparent material so that models can be viewed with little distortion.

The longer side of the test section should lie horizontally,

so that, with the force balance mounted below the test section, half models, components, aerofoils and floor mounted models can be easily tested.

2.5.2 Corner Fillets

Test section corner fillets are recommended for three reasons:

1. Tapered corner fillets can be used to correct for the boundary layer growth in the test section which causes a horizontal pressure gradient (horizontal buoyancy). The exact taper required can be determined when the tunnel is in operation and the rate of boundary layer growth has been determined.
2. Corner fillets eliminate 90 degree corners in the test section and, hence, diminish the secondary flow. Secondary flow is the result of the flow attempting to equalize the velocity gradients at the walls; to accomplish this the air must move into the corners from the region of the higher speed "core" flow.
3. Fillets provide a convenient location for the installation of lights. The fillet size can be determined from this. A fillet hypotenuse of eight inches would be adequate.

CHAPTER III

THE RETURN CIRCUIT

3.1 Introduction

Having chosen the closed return type of wind tunnel, it remains for the designer to satisfy the requirements of space, efficiency, and performance. The problem reduces to providing a return circuit which expands by the highest possible ratio within the space given, and in which the total pressure loss is a minimum.

The largest pressure losses in the return circuit of a conventional tunnel occur at the corners. These losses are proportional to the square of the mean flow speed; therefore the corner losses can be reduced substantially by reducing the flow speed in the return passage as quickly as possible by means of rapid diffusion.

Much of the remainder of this design study consists of a discussion of the design of a wind tunnel incorporating wide angle diffusers with boundary layer control. The decision to proceed with the design of such a wind tunnel was made on the basis of the results of a study of several conventional wind tunnel layouts. The results of this study are summarized in the following tables.

It is evident from the results that the best conventional tunnel configuration is one with a contraction ratio of about six, because such a tunnel fits into the space provided with the highest possible

Contraction Ratio	Test Section Area	Fan Diameter	Maximum Speed	Overall Length
4	21 1/8 ft. ²	7.34 ft.	250 fps	56 ft.
6	21 1/8 ft. ²	8.00 ft.	250 fps	63 ft.
9	21 1/8 ft. ²	8.00 ft.	250 fps	86 ft.
12	21 1/8 ft. ²	8.00 ft.	250 fps	106 ft.

Resistance Coefficients for a Wind Tunnel with
Contraction Ratio of Six (Energy Ratio = 2.82)

Test Section	0.023
Contraction	0.0143
Corner One	0.0264
Corner Two	0.0264
Corner Three	0.0042
Corner Four	0.0042
Fan Assembly	0.0115
Diffuser One	0.15
Diffuser Two	0.027
Model	0.029
Constant Area Ducts	0.0041
Total	0.319

contraction ratio. However the fan motor power required for such a tunnel is about 235 H.P. The decision to proceed with a tunnel using wide angle diffusers was based on the idea that the power required to operate a tunnel with a contraction ratio of six could be substantially reduced by the use of wide angle diffusers with boundary layer control.

The calculations for the preceding results were based on a conventional tunnel layout. It was assumed that the fan was located between the second and the third corners, that the first diffuser was located before the first corner, that the second diffuser lay between the fan and the third corner, and that the equivalent cone diffuser expansion angles should not exceed eight degrees. The method used to calculate the energy ratio was similar to that used in Chapter VI. The diffuser pressure recovery efficiency was taken as 85 percent for both diffusers. (see Appendix A)

The motivation for this design study, in part, grew out of the idea that perhaps some of the research work done in the area of wide angle diffusers could be used in the design of a short, efficient wind tunnel. A subsequent review of the available literature showed that considerable work had been done in the area of diffusers. The applicability of the results of this work to wind tunnel design remains to be determined.

A review of literature indicates that very little has been published concerning the design of the contraction, the corners, or the fan. These wind tunnel components along with the diffusers will be dis-

cussed in the following sections.

3.2 The Contraction

3.2.1 Flow Uniformity

The function of the contraction section is to accelerate the low-speed air entering the settling chamber. The effect of the contraction is to decrease the velocity nonuniformity in the test section. The velocity variation is inversely proportional to the square of the contraction ratio. (see references [1] and [2]) However, reduction in nonuniformity can be accomplished by the use of flow control screens in the settling chamber of a wind tunnel of moderate contraction ratio. In comparison with the use of a large contraction ratio of the order of 16:1, a moderate contraction ratio results in a substantial saving in tunnel length at the expense of some increase in pressure loss in the screens.

3.2.2 Turbulence

Dryden and Schubauer [9] show rather conclusively that the effect of the contraction on isotropic turbulence in the settling chamber is to reduce the longitudinal turbulence component and to increase the lateral components. The results of tests conducted in the National Bureau of Standards 4-1/2 foot wind tunnel show that without screens in the settling chamber the average turbulence in the test section at several speeds was approximately equal to the average turbulence in the settling chamber. However, the turbulence decay formulae predict con-

siderable decay during the time of passage through the contraction. It was concluded that the contraction acts to increase the turbulent energy. (see also reference [10])

It is clear that the ratio of the mean turbulence intensity to the mean velocity does decrease through the contraction because the mean velocity increases substantially. Following the development of Dryden and Schubauer [9], the mean turbulence intensity U' increases by the factor $(\frac{2}{3}n + \frac{1}{3}n^{-2})^{1/2}$ from the settling chamber to the test section where $U' = [\frac{1}{3}(u'^2 + v'^2 + w'^2)]^{1/2}$ u' , v' , and w' are the root mean square values of the velocity fluctuation components, and n is the contraction ratio. Because the mean velocity in the test section is about n times as large as the mean velocity in the settling chamber, the ratio of the mean turbulence intensity to the mean velocity in the test section is about $(\frac{2}{3}n^{-1} + \frac{1}{3}n^{-4})^{1/2}$ its value in the settling chamber. The development ignores turbulent decay, therefore the calculated factor for any given n will be high.

Examining the relation $(\frac{2}{3}n^{-1} + \frac{1}{3}n^{-4})^{1/2}$, it is evident that an increase in the contraction ratio from, for example, five to 15 decreases the preceding by a factor of about $(1/3)^{1/2}$. This is not a substantial decrease when weighed against the great increase in tunnel size.

Low turbulence levels can be achieved in wind tunnels with large contraction ratios and few screens because the return circuit is very long and any turbulence generated in the test section or by the

fan will have a long time to decay before reaching the settling chamber. However, turbulence can be reduced to satisfactory levels in wind tunnels with moderate contraction ratios (five to ten) by the use of more screens with a moderate increase in the power required to operate the tunnel.

3.2.3 Design Discussion

The cross sectional shape of the contraction is determined by the shape of the settling chamber and the shape of the test section. The components of the return passage should be rectangular in cross section to make the best use of the available space. In this way it may be possible to use parts of the walls of the room as walls of the wind tunnel. We must use a contraction section of rectangular cross section because we have chosen a rectangular test section. A rectangular contraction is easy to construct and lends itself to the use of wood in construction.

No analytical method is available for adapting axisymmetric designs to nonaxisymmetric contractions. However, good results have apparently been achieved by making the axial distribution of area in a nonaxisymmetric contraction the same as the axial distribution of area of a good axisymmetric design. Livesey [11] recommends a design by M.J. Cohen and N.J.B. Ritchie, which is similar to the method of H.S. Tsien as applied by B. Szezeniewski. It is suitable as a practical design method.

Although significant flow separation will probably not occur in the contraction section, the boundary layer is more prone to sepa-

rate in the corners of a rectangular cross section contraction because the curvature of the contraction is greatest along the diagonals. The probability of separation can be decreased by extending the test section fillets back into the contraction. This will reduce the corner boundary layer thickness and the curvature on the diagonals.

3.3 The Settling Chamber

The purpose of the settling chamber is to provide a sufficient distance for the flow to straighten and become more uniform after passing through the last corner. The settling chamber provides an ideal location for the installation of honeycombs and screens because of the low flow speed.

The necessity for screens and honeycombs, the possible size of the screens and honeycombs, and the length of the settling chamber are determined by the flow history upstream of the settling chamber. One cannot state specifically what will be required in terms of screens and honeycombs before the tunnel is constructed because it is virtually impossible to predict the type of flow in the settling chamber.

Honeycombs should not be required, because corner vanes serve the same purpose in removing swirl; screens are more effective in removing nonuniformities and large scale turbulence. Since screen effectiveness can be determined quite accurately (see reference [12]), it is best to match the screen or screen combination to the job that must be done after the tunnel is built and operating. The best solution is to provide for the insertion of six screens at eight-inch spacing. The screens can

be mounted on frames outside of the tunnel and inserted into the settling chamber when they are needed.

3.4 Corner Vanes

Corner vanes are necessary in closed return wind tunnels because the corners are rather abrupt right angle turns with only a small radius of curvature. As well as guiding the air around the corners, corner vanes greatly improve flow uniformity. (see reference [1])

Corner vanes prevent the secondary motion that would develop in curved ducts, tending to give the stream a spiral motion following the corner. Without vanes, the greatest pressure loss would occur in the straight section following the corner as the extra kinetic energy is dissipated.

The resistance coefficient of the corner decreases as R/H increases where R is the radius of curvature of the corner and H is the height of the corner. The use of corner vanes increases the ratio R/H .

One other problem associated with corners is the possibility of flow separation due to the adverse pressure gradient preceding the corner entrance. The pressure rises toward the outside of the corner due to the centrifugal force. The flow entering the corner near the outside edge faces a severe adverse pressure gradient which may cause the boundary layer to separate. The problem is alleviated by a reduction in the flow velocity and an increase in the radius of curvature. Flow expansion through corners is undesirable because the possibility of flow separation is high due to the additional built-in pressure gradient of

the diffuser combining with the adverse pressure gradient already present.

Sheet metal vanes are easier and less costly to construct and may be more efficient than aerofoil section vanes. Thick vanes cause blockage and have adverse pressure gradients over the trailing section which may cause flow separation.

Bradshaw and Pankhurst [3] recommend a vane arrangement with a gap/chord ratio of 1:4, a leading edge angle of attack of four to five degrees, and a trailing edge angle of zero degrees with a total included angle of about 86 degrees. In tests at Reynolds numbers of 2×10^5 , a pressure drop coefficient of 0.06 was achieved neglecting secondary flow losses at the roots. (The pressure drop coefficient is defined as the ratio of the total head loss in passing through the corner to the dynamic head at the corner).

3.5 The Fan

The efficiency of the fan depends upon the flow speed through the fan. Consider the local advance ratio $j = U/qd$ where U is the mean flow velocity at the fan, q is the fan rotational speed in revolutions per second, and d is the diameter at the fan element under consideration. The local air speed must be increased to keep the advance ratio over the entire fan blade within the high efficiency range (one to ten) while maintaining a high rotational speed. Consequently, the higher speed regions of the wind tunnel are more suitable for the fan location. (see reference [2]) This is doubly beneficial because the cost of the fan

is an increasing function of the fan area and the high speed sections of the wind tunnel are the sections with low cross sectional areas.

The fan tip speed should be kept below 0.5 times the speed of sound to avoid compressibility problems. Extremely high rotational speeds should also be avoided because wind tunnel noise is more intense at high frequencies.

One significant advantage of high rotational speed is the accompanying decrease in swirl velocity. The swirl velocity is defined as $w = dp/\rho q r^2$, where dp is the net pressure rise across the fan, ρ is air density, q is the fan rotational speed, and r is the given radius at the element being considered. At a given r , w is inversely proportional to q . The swirl energy is converted into a pressure rise by the use of straightener vanes or by the fan itself if prerotation vanes are used.

The velocity profile preceding the fan is more important in terms of fan vibration than in terms of fan efficiency. Fan vibration is often caused by an asymmetrical velocity front reaching the fan. The fan will tend to remove circumferential variations in mean velocity as well as radial variations and, hence, tend to change the velocity distribution to correspond more closely to the desired velocity distribution. To avoid resonant frequencies, the number of fan blades should not equal the number of straightener vanes.

3.5.1 The Fan Nacelle

The nacelle surrounding the fan hub serves several purposes. In addition to reducing the drag over the hub, the nacelle reduces the

cross sectional area at the fan and, therefore, makes a higher advance ratio possible. At the same time, the nacelle encloses the inefficient thick root sections of the fan blades which also cause the greatest swirl.

However, the nacelle also presents major problems. The length of the nacelle must increase as the boss diameter increases, to provide for proper streamlining. The larger the nacelle, the greater the drag. Bradshaw and Pankhurst [3] recommend a 3:1 length to diameter ratio as a minimum requirement.

To avoid the problem of flow separation over the nacelle tail, the tunnel walls can be shaped to follow the contours of the nacelle, thereby forming a constant area duct. This increases the length of the tunnel because the overall diffuser expansion ratio is increased. In addition, curved sections are costly to construct. A compromise solution would be to contract the outer walls slightly and limit the equivalent cone expansion angle over the nacelle tail to five degrees.

3.5.2 Design Discussion

The fan itself cannot be designed without knowledge of the energy ratio of the wind tunnel or, in other words, without knowing the pressure rise required of the fan. It is doubtful whether the energy ratio can be estimated closely enough to justify a detailed fan design. The cautious designer will underestimate the energy ratio so that the fan will not absorb the maximum motor power at the maximum designed speed. An overestimate of the energy ratio would mean that the air would not

reach the designed speed at full motor power.

The preliminary design principles discussed in the preceding sections are usually sufficient to determine the basic fan dimensions. (A detailed fan design procedure can be found in reference [2]). For our particular wind tunnel, the fan dimensions and the fan location depend on the diffuser arrangement, which will be discussed in Chapter V.

CHAPTER IV

THE DIFFUSERS

4.1 Flow Separation in Diffusers

The design of the diffuser can be approached as an energy conversion problem. Ideally, a diffuser converts kinetic energy into pressure energy (or potential energy). Since air is a real gas, some kinetic energy is also converted into heat through viscous dissipation.

Outside of the diffuser boundary layer, the flow can be treated as potential flow. In this region, fluid particles convert their kinetic energy into pressure energy. However, fluid particles in the boundary layer are slowed by the adverse pressure gradient (impressed on the boundary layer) and also by surface friction. Eventually the boundary layer must come to rest because its kinetic energy is being converted at a faster rate than the rate of energy conversion of the air in the "potential core". The boundary layer flow then reverses direction under the influence of the adverse pressure gradient and causes separation. The process culminates in the formation of large eddies which consume more of the flow energy and reduce the pressure recovery of the diffuser.

Flow separation in diffusers seems to depend on three factors:

1. The inlet momentum boundary layer thickness.
2. The severity of the pressure gradient.
3. The rate of boundary layer growth.

One can assume that the larger the initial boundary layer thickness and the faster the growth rate, the greater the possibility of separation, because of a significant reduction in the effectiveness of momentum transfer to particles deep inside the boundary layer.

Ideally, the total pressure rise in a diffuser is a function of the area ratio and the inlet velocity. The pressure gradient at any given point in the diffuser is a function of the diffuser inlet area, the area ratio, the diffuser length, and the inlet velocity. All other factors being equal, including the divergence angle but excluding the length, the conical diffuser with the larger entrance area will have less severe pressure gradients near the inlet than the diffuser with the smaller inlet area. In the case of two ideal conical diffusers of equal inlet area, inlet velocity, and area ratio, but with different lengths, the longer diffuser will have the more gradual pressure rise and less severe pressure gradients near the inlet. (see Figure 4.1)

Two types of diffusers of interest in wind tunnel design are prone to flow separation: very short diffusers with large area ratios, and very long diffusers with large area ratios. Very short diffusers will exhibit flow separation near the inlet due to the very adverse pressure gradients near the inlet. The flow in the long diffusers will separate near the outlet because the skin friction acting over the excessive length will slow the boundary layer to zero velocity before the outlet.

Wind tunnel designers recommend that diffuser area ratios

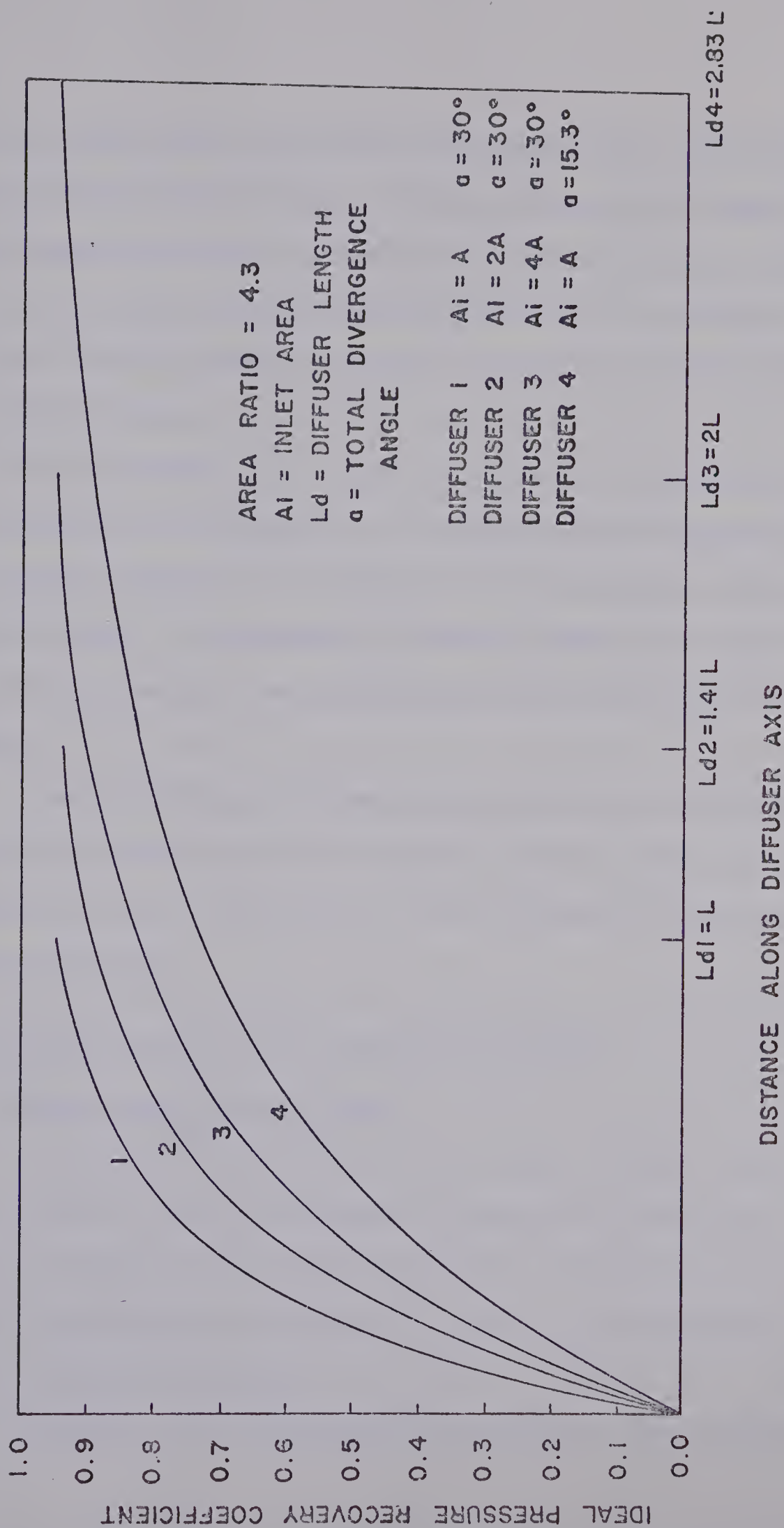


FIGURE 4.1. PRESSURE RISE IN CONICAL DIFFUSERS IN THE AXIAL DIRECTION

should not exceed three, with total included equivalent cone angles of not more than about eight degrees. Within this range, diffusers, with entrance areas that are not extremely small, are long enough so that severe adverse pressure gradients do not occur near the diffuser inlet. At the same time the diffusers are not so long that separation will occur near the outlet.

At the present time no general method of diffuser performance prediction exists for diffusers of all shapes and sizes and with all types of inlet conditions. Our intent is not to design an optimum conventional diffuser. The preceding discussion suggests that rapid diffusion may be possible if the boundary layer can be energized or removed altogether.

Improved diffuser performance may also be possible by proper shaping of the diffuser walls to reduce the adverse pressure gradients. This method has been investigated for the two-dimensional case only. (see reference [16])

4.2 Wide Angle Diffusers with Boundary Layer Control

4.2.1 Boundary Layer Control Methods

The practical application of wide angle diffusers necessitates the application of special techniques to prevent flow separation. In addition to boundary layer suction and blowing, the other proven methods of separation prevention are the use of screens, vortex generators, windmills, and splitter vanes.

Windmills and vortex generators essentially act as pumps since

they cause the air to move from regions of high dynamic pressure to regions of low dynamic pressure. Vortex generators are not effective in very wide angle diffusers, while the use of windmills often results in swirl and relatively high pressure losses.

Screens act to smooth the velocity profile and, therefore, perform essentially the same function as windmills, although much more effectively. Pressure losses associated with screens are high.

Splitter vanes prevent flow separation by splitting a wide angle passage into several smaller angle passages. They have only been used successfully in two-dimensional diffusers. (See reference [17])

Blowing and suction have been used to prevent flow separation in diffusers with divergence angles of up to 50 degrees. In the following sections we will consider these methods of flow separation prevention, because it is clear that the other methods are not suitable for our purpose.

4.2.2 Design Problems with Boundary Layer Control by Suction and Blowing

Research in the area of diffuser boundary layer control seems to be scattered and poorly co-ordinated. Much of the research has been prompted by the need to solve particular isolated problems.

The wind tunnel designer faces three problems in the design of wide angle diffusers with boundary layer control that the researchers have either ignored or controlled:

1. The shape of a wind tunnel diffuser is usually not a simple cone or rectangular two dimensional expansion, but is usually a transition

section such as rectangular to round.

2. The momentum boundary layer thickness at the diffuser inlet varies with the flow velocity in the wind tunnel.

3. The effect of model wake on diffuser performance must be considered.

There are four factors to consider in the application of boundary layer control information on wide angle diffusers: the possible scale effects, the transformation efficiency, the stability of the flow, and the power required to drive the boundary layer control system.

In general, the results of small scale tests are conservative because the ratio of boundary layer thickness to diffuser inlet diameter can be expected to be smaller for the full size case than for the model.

Transformation efficiency is defined as the ratio of the actual pressure rise coefficient to the theoretical pressure rise coefficient. Diffusers with low transformation efficiencies are of little value because low efficiency implies the existence of considerable flow separation. This means that the "core" velocity at the diffuser outlet is relatively high. In our application, rapid efficient diffusion is necessary if large corner pressure losses are to be avoided.

An effective boundary layer control system gives transformation efficiencies near 100 percent and good flow stability under extreme conditions. Given an effective system, the cost to operate the system must be weighed against the advantages gained in terms of space and reduced power losses in other parts of the tunnel.

4.2.3 Flow Stability and Diffuser Performance

Studies of value to the wind tunnel designer should deal with the problem of predicting and maintaining flow stability as well as the problem of deriving some design principles for the design of wide angle diffusers with boundary layer control. Studies by Moon [13] and Savage [14] have directly or indirectly touched upon some of the problems associated with the application of research findings to actual boundary layer control diffuser design.

Moon [13] describes three possible flow conditions in the diffuser: inoperable, unstable, and stable. The inoperable case is characterized by complete flow separation. In the unstable state, the flow detachs when a slight disturbance or fluctuation is introduced and does not reattach when the disturbance is removed. In general, the flow moves from one condition to the next with an increase in the ratio of suction flow quantity to the main flow quantity.

Moon found that flow turbulence was the main cause of flow instability even when boundary layer suction was used. In the wind tunnel it is likely that the flow at some time will be greatly disturbed, particularly in the area immediately downstream of the test section. Moon's study suggests that under these conditions the performance of the diffuser downstream of the test section can be maintained at a reasonable level by the use of boundary layer suction upstream of the diffuser inlet which reduces the inlet boundary layer momentum thickness. The performance of a small angle diffuser will decrease when highly disturbed

flow exists at the diffuser inlet (see reference [14]).

High turbulence conditions will exist at the diffuser inlet very infrequently. Under ordinary testing conditions (small model wakes), the length of the test section will allow sufficient flow mixing to occur so that the flow entering the diffuser will be stable and uniform and the diffuser efficiency will be maintained.

4.3 The Fan and the Diffusers

A rapid area expansion between the test section and the first corner, although necessary to reduce the air speed in the corner, eliminates the possibility of locating the fan downstream of the second corner. For example, if the expansion ratio to the first corner is 6:1 and there is no further expansion to the fan, calculations show that the fan would have to be 13.3 feet in diameter and operate at a speed of 630 revolutions per minute to give a maximum test section speed of about 250 feet per second. This size of fan and fan speed are undesirable, therefore the fan must lie between the test section and the first corner even though this location is undesirable from the point of view of noise and is susceptible to damage if the model should break up.

Dryden and Schubauer [9] present experimental results for a wind tunnel with a fan in the position described above, showing that the fan noise is readily transmitted to the test section and is responsible for some test section turbulence. The sound turbulence intensity was found to be 0.028 percent. They further suggest that the addition of

screens in the settling chamber actually contributes to an increase in the base sound turbulence level. The addition of each screen forces the fan to run faster to make up for the pressure loss through the screen. Assuming that the noise level increases with increasing fan speed, it follows that the turbulence intensity due to sound should increase also. Dryden and Schubauer were not able to reduce the effect with a combination of screens and silk bolting cloth. However, a turbulence level of the order of a few hundredths of one percent is satisfactory for most wind tunnel testing.

A fan in the position before the first corner is vulnerable to damage by model fragments should failure occur. In the position following the second corner, the fan would be protected by the corner vanes and the distance between the test section and the fan. For our case it would be wise to place a heavy mesh screen in front of the fan.

CHAPTER V

DIFFUSER, CONTRACTION, AND SETTLING CHAMBER DESIGN

5.1 The Diffusers

An arrangement consisting of two diffusers linked by the fan duct is necessary in order to achieve a large expansion to the first corner while at the same time keeping the fan area small.

In the design of the diffusers, the work of Savage [14] is of interest for two reasons:

1. One diffuser section tested was a transition section from rectangular to round.
2. Tests were conducted with a bluff body (a cylinder) placed in the test section.

Savage's work was initiated by the need to design a short diffuser for an open circuit wind tunnel. Air was drawn in through slots at the diffuser inlet.

Savage's results show the detrimental effect of the bluff body in the test section. For a five degree equivalent divergence angle transition diffuser, the effect of the bluff body was to decrease the diffuser transformation efficiency from 78 percent to 52 percent. With the same bluff body, the transformation efficiency of the slotted transition diffuser (total equivalent cone angle of 17.6 degrees) decreased from a maximum of 69 percent to about 46 percent.

It had been thought that it might be possible to use blowing on the first diffuser and suction on the second diffuser. This would make it possible to link the systems and use the natural pressure difference to drive the system. Although Savage achieved a considerable improvement in the performance of a wide angle transition diffuser by using blowing at the diffuser inlet, the diffuser efficiency obtained is not acceptable for our purpose and therefore the idea of a system of suction and blowing will not be pursued.

Moon [13] used distributed suction boundary layer control and achieved a pressure recovery of approximately 100 percent by means of distributed suction on a conical diffuser of 30 degrees total included angle.

The test arrangement of apparatus used by Moon is more favorable than the arrangement that can be expected in the wind tunnel. In Moon's apparatus, the fan duct walls followed the fan nacelle profile; following the fan assembly was a constant area duct with cross sectional area less than the fan flow area and of length about twice the fan diameter. The diffuser followed this duct. Moon used boundary layer suction and axial blowing on the fan nacelle to eliminate a central core region of backflow or stagnant air in the diffuser, because extending the fan duct beyond the tip of the nacelle was only slightly effective in smoothing the velocity profile before the diffuser inlet. The turbulence created by the axial blowing was reduced by the addition of two screens that caused a pressure loss of 0.7 times the dynamic head at

the screens.

The preceding suggests that boundary layer suction on the nacelle may be necessary to achieve a uniform velocity profile at the inlet to the diffuser following the fan. To avoid the use of axial blowing and the need for screens with high pressure losses, the area of boundary layer suction on the second diffuser should be extended into the fan duct to reduce the inlet momentum boundary layer thickness.

By making a conservative choice of area ratio and divergence angle, we may eliminate the danger in applying the results of conical diffuser studies to the design of transition section diffusers. Both diffusers will have an equivalent cone angle of 30 degrees and use suction boundary layer control. The area expansion ratio of the first diffuser will be 2:1, while that of the second diffuser will be 2.5:1. These are both conservative ratios. The expansion of 2:1 is in keeping with the suggestion of Bradshaw and Pankhurst [3]; it is less than the expansion from the fan to the first corner because of the greater separation risk following the test section. The lengths of the diffusers shown in Figure 5.1 were determined by simple calculations, knowing the area and the equivalent diameters of the test section, the fan, and the first corner.

5.2 The Fan Assembly

In determining the length of the fan assembly, the diameter of the fan nacelle and the space required for screen installation must be taken into account. Because the fan duct will converge only slightly,

the expansion over the nacelle trailing portion must be considered. Bradshaw and Pankhurst [3] suggest that the maximum expansion be equivalent to a conical expansion with five degrees total divergence angle. This, presumably, ensures attached flow over the nacelle and the surrounding duct wall. Using a maximum nacelle diameter of 0.3 times the fan duct diameter and assuming that the nacelle trailing length is one half of the total length, the approximate nacelle length can be determined. The overall fan duct length of 9.0 feet is equal to the nacelle length plus an additional 0.5 feet for screen installation.

5.3 The Tunnel Layout

The overall length of the wind tunnel is 71 feet. The first corner is of square cross section; the dimensions are easily determined.

We will allow 1.6 times the equivalent inlet diameter for the length of the settling chamber and the contraction section. The dimensions of the settling chamber inlet were found by a trial and error solution where an attempt was made to keep the curvature of all sides of the contraction equal. The equivalent expansion angle of the return diffuser (number three) is much less than 5 degrees and the expansion ratio is 1.14; hence, there is no danger of flow separation in this diffuser. The calculated lengths of the tunnel components are given below.

<u>Component</u>	<u>Length</u>
Corner Four	10.3 feet
Settling Chamber Contraction	19.6 feet
Test Section	10.0 feet
Diffuser One	4.0 feet
Diffuser Two	7.9 feet
Diffuser Three	50.5 feet
Fan Duct	9.0 feet
Corner One	10.2 feet

Figure 5.1 is the draft layout of the wind tunnel.

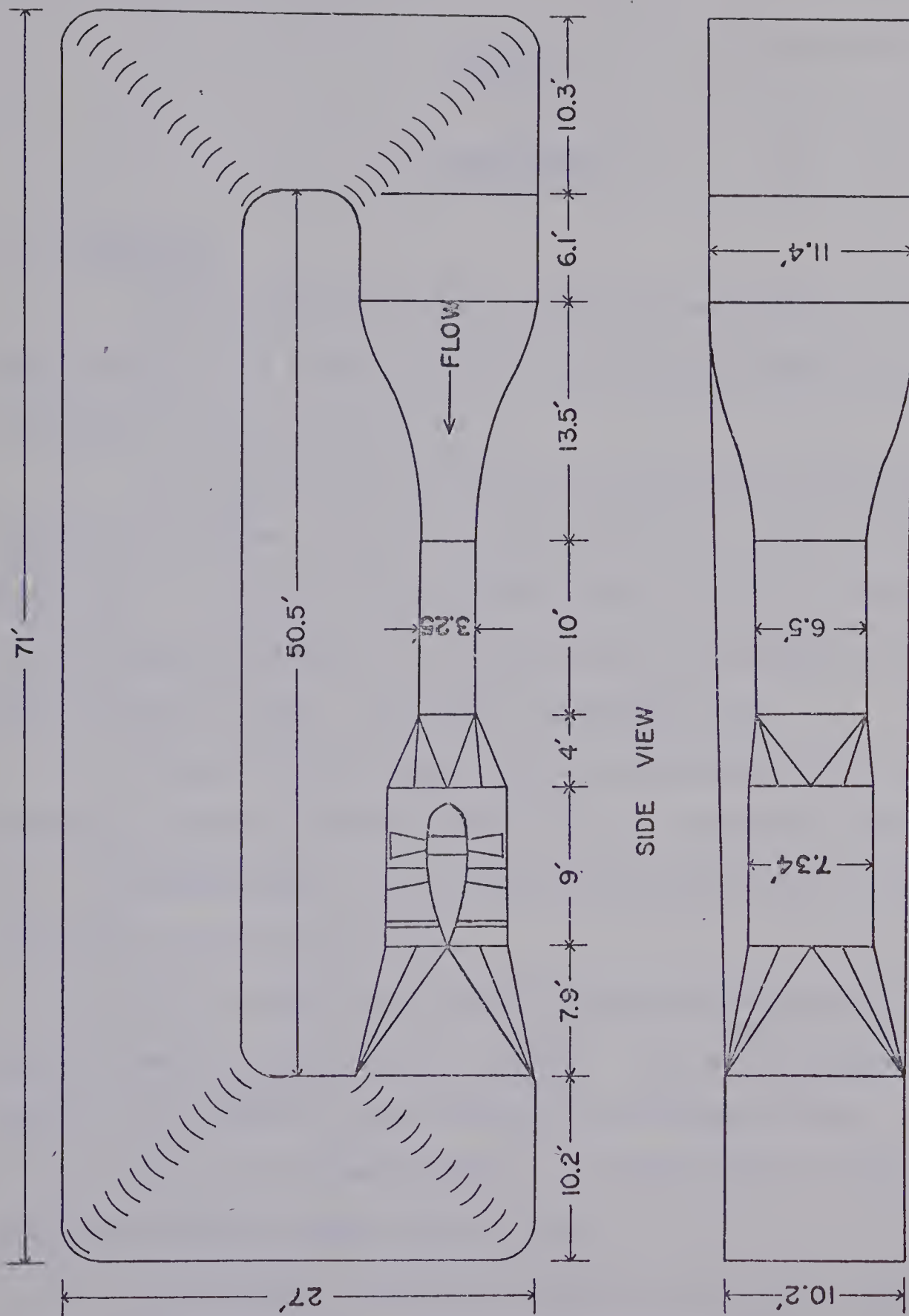


FIGURE 5.1 WIND TUNNEL LAYOUT

CHAPTER VI

CALCULATIONS

6.1 Discussion

Ideally, the wind tunnel calculations determine the fan power required, the pressure loading on the tunnel components, and the wind tunnel operating temperature.

The following calculations follow the method outlined by Gorlin and Slezinger [1] and to a lesser extent the outline of Pope and Harper [2]. The total resistance coefficient of each wind tunnel component is calculated separately, using the empirical or semiempirical relations that have been derived. The general assumptions are:

1. Compressibility effects can be ignored because the flow velocity in the wind tunnel is much less than the speed of sound.
2. Energy losses in the airstream are mainly due to frictional resistance and local resistance.
3. The frictional resistance is assumed to depend mainly on the Reynolds number and the surface roughness. The local resistance is assumed to be a result of flow separation and turbulent mixing.
4. The air is assumed to flow in a uniform manner through sections of constant cross sectional area.
5. The effects of the Reynolds number and the Mach number are assumed to be small at normal wind tunnel velocities.

6.2 The Resistance Coefficient

The total hydraulic resistance coefficient can be determined for each wind tunnel component. These coefficients can be expressed in terms of the velocity head in the test section by multiplying the coefficient by the square of the ratio of the test section area to the tunnel component area.

The greatest loss in the tunnel due to skin friction occurs at the highest tunnel velocity, since the frictional resistance is proportional to the square of the speed. Gorlin and Slezinger [1] use the relative geometrical roughness of the material and the flow Reynolds number (based on the hydraulic diameter) to determine the skin friction coefficient. Following this method, we find that for typical construction materials, such as steel or plywood the average skin friction coefficient is about 0.010. This value will be used in the calculations.

6.3 Energy Losses

6.3.1 The Test Section

$$L = 10 \text{ feet}$$

$$a = 3.25 \text{ feet}$$

$$b = 6.50 \text{ feet}$$

$$D_h = 2ab/(a + b) = 4.34 \text{ feet}$$

$$K_{ts} = K = \lambda L/D_h = \underline{0.023}$$

6.3.2 The Contraction and the Settling Chamber

Assuming that the losses in the contraction are mainly due to friction, for a contraction of rectangular section:

$$n = 5.72$$

$$L = 13.5 \text{ feet}$$

$$D_h = 4.34 \text{ feet}$$

$$K_{ts} = \frac{4}{9} (\lambda L / D_h) = \underline{0.0138}$$

The losses in the settling chamber (excluding screen losses) are those of a straight duct.

$$L = 6.1 \text{ feet}$$

$$a = 11.5 \text{ feet}$$

$$b = 10.3 \text{ feet}$$

$$A_{ts}/A = 1/5.72$$

$$D_h = 2ab/(a + b) = 10.85 \text{ feet}$$

$$K_{td} = (\lambda L / D_h) (A_{ts}/A)^2 = \underline{0.00017}$$

The resistance coefficient of the settling chamber is approximately equal to 1/100 the resistance coefficient of the contraction section. The straight sections between corners one and two and corners three and four are of about the same size as the settling chamber and, therefore, also have negligible resistance coefficients.

6.3.3 The Corners

Using K equal to 0.15 for all four corners, we find that for corners one and two:

$$A_{ts}/A = 1/5$$

$$K_{ts} = (A_{ts}/A)^2(0.15) = \underline{0.006}$$

and for corners three and four:

$$A_{ts}/A = 1/5.72$$

$$K_{ts} = (A_{ts}/A)^2(0.15) = \underline{0.00458}$$

6.3.4 The Third Diffuser

The third diffuser can be treated as a straight duct, because the expansion ratio is only 1.14:1.

$$L = 50.5 \text{ feet}$$

$$a = 10.2 \text{ feet}$$

$$b = 10.9 \text{ feet}$$

$$A_{ts}/A = 1/5.36$$

$$D_h = 2ab/(a + b) = 10.55 \text{ feet}$$

$$K_{ts} = \underline{0.00167}$$

6.3.5 The Fan Assembly

The resistance of the fan assembly includes the resistance of the nacelle, the fan duct, and the safety screen in front of the fan.

Therefore:

$$K = K_n + K_d + K_s$$

where:

$$K_n = (C_{xf} A_{nf}/A_{ff}) / (1 - A_{nf}/A_{ff})^3$$

$$K_d = \lambda L/D_f$$

$$K_s = 0.02$$

$$C_{xf} = 0.25$$

$$A_{nf} = 0.09A_f$$

$$A_{ff} = 0.91A_f$$

$$L = 9 \text{ feet}$$

$$D_f = 7.26 \text{ feet}$$

$$K = 0.0664$$

$$K_{ts} = \underline{0.02}$$

6.3.6 The Diffusers

The power required to drive the boundary layer control system can be determined from the suction mass flow rate and the pumping pressure coefficient. Assuming a pumping efficiency of 75 percent, the pumping power is equal to $P_s Q_s / 0.75$ where P_s is the suction pressure and Q_s is the suction flow quantity. For the purpose of comparison, the ratio of the pumping power to inlet flow power is used. This ratio is $P_s Q_s / 0.75 q_1 Q_1$ where q_1 is the inlet dynamic head and Q_1 is the inlet flow quantity.

Holtzhauser and Hall [15] found P_s equal to $0.7q_1$ and Q_s/Q_1 equal to 0.03, with a suction length 0.16 times the diffuser length. These figures represent the minimum quantities at which separation was shown to be eliminated. A 30 degree conical diffuser of area ratio 2:1 with an inlet Mach number of 0.2 was tested and a pressure recovery of nearly 100 percent was achieved. The power ratio was 0.028.

Moon [13] achieved a pressure recovery of nearly 100 percent for several combinations of screens and honeycombs on a 30 degree conical

diffuser with distributed suction over the whole area. By extending the suction area back into the region upstream of the diffuser, Moon was able to remove the screens and maintain the diffuser performance. Mass flows of six to eight percent of the main flow quantity and an average suction pressure of about 1.7 times the inlet dynamic head were required. The power ratio for the tests was about 0.16, where the mean inlet velocity was about 60 feet per second.

Assuming that the results of Holtzhauser and Hall can be duplicated, the power ratio for suction should not be greater than 0.05. Therefore, we can assume that the power required to run the boundary layer control suction for each diffuser should probably be no more than 0.05 times the flow power at the diffuser inlet, if suction is applied over only a fraction of the diffuser surface area.

For the purpose of calculating the energy ratio, this power can be expressed in resistance coefficient form. For diffuser one K_{ts} is equal to 0.05, and for diffuser two K_{ts} is equal to 0.0125; the dynamic head at the inlet to diffuser two is 0.25 times the dynamic head at the inlet to diffuser one.

The resistance coefficient of the first diffuser can be expected to increase if the diffuser becomes partially stalled as a result of a large turbulent wake entering the diffuser from the test section. This should occur very infrequently.

6.3.7 The Resistance of a Model

As an extreme case of model drag, consider a model of span

0.8 times the test section width, aspect ratio equal to five, and drag coefficient equal to 0.5. In our wind tunnel, the largest models to be tested at high speeds will likely be aircraft component sections, whose largest median area should not exceed five percent of the test section area, and whose drag coefficient should be less than 0.5. Therefore:

$K = (C_x A_m / A_{ts}) / (1 - A_m / A_{ts})^3$, where C_x is equal to 0.5 and A_m / A_{ts} is equal to 0.05, and $K_{ts} = K = \underline{0.029}$.

6.3.8 The Energy Ratio

Following is a summary of the tunnel resistance coefficients:

<u>Component</u>	<u>K_{ts}</u>
Test Section	0.023
Contraction	0.014
Corner One	0.006
Corner Two	0.006
Corner Three	0.0046
Corner Four	0.0046
Fan Assembly	0.02
Diffuser One	0.05
Diffuser Two	0.0125
Diffuser Three	0.0017
Model	<u>0.029</u>
Total	0.171

The total resistance coefficient should be increased by about ten percent to account for pressure losses due to screens in the settling chamber and the power required for boundary layer control on the fan nacelle. Excluding the loss due to the model (for the purpose of comparison with other studies), the total resistance coefficient becomes 0.156 and the energy ratio is 5.76. The tunnel power required for this energy ratio is 115 H.P.

6.4 Wind Tunnel Heating

The temperature rise incurred in bringing the air in the wind tunnel to rest from a test section speed of 250 feet per second is 5.2°F . For an energy ratio of 5.76, the temperature rise (without cooling) per wind tunnel circuit would be about 0.9°F . (see following calculations)

The air stream can be cooled by massive air exchange, internal heat exchange with radiators, and surface heat exchange with running water. However, special cooling should not be necessary. The following calculations show that the air exchange due to boundary layer suction alone should be sufficient to limit the final operating temperature of the air stream at the maximum speed to 20°F above the room temperature.

Calculations

$$\text{Air Stream Power} = \frac{1}{2} \rho V^3 A_{ts} = RgC_p V A \, dT$$

$$\text{so } dT = \text{temperature rise} = \frac{V^2}{2gCR}$$

$$\text{where } C = 0.24 \text{ BTU/lbm}^{\circ}\text{F}$$

$$g = 32.17 \text{ lbm/slug}$$

$$R = 778 \text{ ft-lbf/BTU}$$

$$V = 250 \text{ ft/second}$$

therefore $dT = 5.2^\circ\text{F}$.

The temperature rise per circuit = $\Delta T = \frac{dT}{ER}$, where $ER = \text{energy ratio} = 5.76$, so $\Delta T = 0.9^\circ\text{F}$. Assume that 5% of the air is extracted per circuit by boundary layer suction and replaced with room temperature air. The tunnel air temperature will rise until $T = 0.95 (T + \Delta T)$, or until $T = 19\Delta T = 17.1^\circ\text{F}$, where T is the increase in the tunnel temperature above room temperature.

CHAPTER VII

CONCLUSIONS

7.1 Summary

Following in point form is a summary of the results of the design study.

1. A wind tunnel using boundary layer control wide angle diffusers has been designed with about half the pressure loss of an equivalent tunnel of more conventional design.

2. The energy ratio of the wind tunnel is greater than five primarily because of the rapid 5:1 area expansion to the first corner made possible by the use of wide angle diffusers with boundary layer control.

3. The regular test section of this wind tunnel is suitable for testing aerofoil sections, half models, component sections, and industrial models.

7.2 Remarks

The wind tunnel that has been studied (contraction ratio of 5.72) is superior to the conventional wind tunnel with a contraction ratio of six because of the higher energy ratio. The increase in energy ratio from 2.82 to 5.76 results in a decrease in the power required from 235 H.P. to 115 H.P. and a subsequent saving in the initial motor cost. The actual fan motor power required will be less than 115 H.P. if sepa-

rate motors are used for the boundary layer control system.

The tunnel design chosen has a third diffuser of low divergence angle and low area ratio that can be replaced by a large test section and a diffuser with a slightly higher divergence angle. This test section would be useful for testing large models at low speeds provided the test section can be made accessible.

The location of the wind tunnel fan immediately downstream of the test section is not common in closed return wind tunnel design. However, for this particular design, the fan must lie in this position. It is interesting to note that most open return tunnels have the fan located in the same relative position although somewhat further downstream. Induced turbulence in the test section, due to fan noise, should not be a problem if the fan is well designed.

The wide angle diffusers should perform satisfactorily if wise use is made of suction boundary layer control.

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APPENDIX A

DESIGN CALCULATIONS FOR A CONVENTIONAL WIND TUNNELWITH A CONTRACTION RATIO OF 6:1A. Wind Tunnel DimensionsTest Section

Width = 6.50 ft.

Depth = 3.25 ft. Equivalent Diameter = $\left(\frac{4 \times 21.125}{\pi}\right)^{1/2} = \underline{5.19}$ ft.Length = 10 ft.Area = 21.125 ft.²Contraction and Settling Chamber

Contraction Ratio = 6

Area = 6 x 21.125 = 126.75 ft.²Equivalent Diameter = $\left(\frac{4 \times 126.75}{\pi}\right)^{1/2} = \underline{12.7}$ ft.Length of Contraction and Settling Chamber = 1.6 x Equivalent Diameter
= 20.4 ft.Corner Four

To make the curvature along the diagonals of the contraction equal:

$$d_4 - 3.25 = w_4 - 6.50 \quad \text{where } d_4 = \text{corner depth}$$

$$w_4 = \text{corner width}$$

and $w_4 d_4 = 126.75 \text{ ft.}^2$

Solving these equations: $d_4 = \underline{9.75 \text{ ft.}}$

$w_4 = 13 \text{ ft.}$

Fan Assembly and Diffusers

We will allow at least 14 feet for the length of the fan duct and a transition section from the second corner to the fan duct (rectangular to round transition).

A_f = fan duct area

A_c = area at corners one and two

X = distance between corners four and one (and corners two and three)

D_{c1} = equivalent diameter at corners one and two

D_{c4} = equivalent diameter at corners three and four = 12.7 ft.

L_1 = length of diffuser one = $X - (20.4 + 10) = X - 30.4$

L_2 = length of diffuser two = $X - 14$

Equivalent cone expansion angle for both diffusers = 8°

Fan diameter = $D_f = D_{c1}$

D_{ts} = equivalent diameter of test section = 5.19 feet

For the expansion angles of both diffusers to be equal:

$$\frac{D_{c1} - D_{ts}}{2L_1} = \frac{D_{c4} - D_f}{2L_2}$$

but

$$\frac{D_{c1} - D_{ts}}{2L_1} = \frac{D_f - D_{ts}}{2(X - 30.4)} = \tan 4^\circ \quad (1)$$

and

$$\frac{D_{c4} - D_f}{2L_2} = \frac{D_{c4} - D_f}{2(X - 14)} = \tan 4^\circ \quad (2)$$

so

$$X = \frac{D_{c4} - D_f}{2 \tan 4^\circ} + 14 = \frac{12.7 - D_f}{2 \tan 4^\circ} + 14 = 105 - 7.17 D_f$$

Substituting for X in equation (1), we find:

$$D_f = 7.8 \text{ feet}$$

Setting $D_f = 8.0$ feet for convenience, the length of diffuser two

becomes $\frac{12.7 - 8}{2 \tan 4^\circ} = 33.8$ ft., and the length of diffuser one is then $(33.8 + 14) - (20.4 + 10) = 17.4$ feet.

The divergence angle of diffuser one is now $2 \tan^{-1} \frac{8 - 5.19}{2(17.4)} = 9.17^\circ$.

The first diffuser is a rectangular section diffuser while the second diffuser is a transition section from the circular fan duct to the rectangular third corner. To make the two expansion angles of diffuser one equal:

$$d_1 - 3.25 = w_1 - 6.50 \text{ where again } d_1 = \text{corner depth (corner one)}$$

$$w_1 = \text{corner width (corner one)}$$

$$\text{and } w_1 d_1 = A_f = \frac{\pi(8)^2}{4} = 50.3 \text{ ft.}^2$$

Solving these equations we find $d_1 = 5.65$ feet

$$w_1 = 8.9 \text{ feet}$$

Tunnel Length

The minimum total tunnel length is equal to L_T where

$$L_T = \text{depth of corner four} + \text{length of settling chamber and}$$

contraction + length of test section + length of diffuser

one + depth of corner one

$$= 9.75 \text{ ft.} + 20.4 \text{ ft.} + 10 \text{ ft.} + 17.4 \text{ ft.} + 5.65 \text{ ft.}$$

$$L_T = \underline{63.2} \text{ feet}$$

B. Resistance Coefficients ($\lambda = 0.010$)

(i) Test Section

$$L = 10 \text{ ft.}$$

$$a = 3.25 \text{ ft.}$$

$$b = 6.50 \text{ ft.}$$

$$D_h = \frac{2ab}{a+b} = 4.34 \text{ ft.}$$

$$K_{ts} = \frac{\lambda L}{D_h} = \underline{0.023}$$

(ii) Contraction

$$L = 14 \text{ ft.}$$

$$D_h = 4.34 \text{ ft.}$$

$$K_{ts} = \frac{4\lambda L}{9D_h} = \underline{0.0143}$$

(iii) Corners One and Two

$$\frac{A}{A_{ts}} = 2.38$$

$$\text{Assume } K = 0.15$$

$$K_{ts} = \left(\frac{1}{2.38}\right)^2 K = \underline{0.0264}$$

(iv) Corners Three and Four

$$\frac{A}{A_{ts}} = 6$$

$$\text{Assume } K = 0.15$$

$$K_{ts} = \left(\frac{1}{6}\right)^2 K = \underline{0.0042}$$

(v) Diffuser One

Assume the diffuser pressure recovery is 85 percent.

$$K_{ts} = \underline{0.15}$$

(vi) Diffuser Two

Assume the diffuser pressure recovery is 85 percent.

$$K_{ts} = \left(\frac{1}{2.38}\right)^2 (0.15) = \underline{0.027}$$

(vii) Constant Area Ducts

$$L = 16.2 \text{ ft.}$$

$$a = 5.64 \text{ ft.}$$

$$b = 8.89 \text{ ft.}$$

$$D_h = \frac{2ab}{a+b} = 6.92 \text{ ft.}$$

$$K = \frac{\lambda L}{D_h} = 0.0234$$

$$K_{ts} = \left(\frac{1}{2.38}\right)^2 K = \underline{0.0041}$$

(viii) Fan Assembly

$$\text{Fan Diameter} = D_f = 8.0 \text{ ft.}$$

$$L = 9.0 \text{ ft.}$$

$$K_s = 0.02$$

$$K_d = \frac{\lambda L}{D_f} = 0.011$$

$$K_n = \frac{C_{xf} A_{nf}}{A_{ff} \left(1 - \frac{A_{nf}}{A_{ff}}\right)^3}$$

$$\text{Assume } C_{xf} = 0.25$$

$$A_{nf} = 0.09 A_f$$

$$A_{ff} = 0.91 A_f$$

$$\text{Then } K_n = 0.0339$$

$$K_{\text{total}} = K_n + K_d + K_s = 0.0649$$

$$K_{ts} = \underline{0.0115}$$

(ix) Model

$$\text{Assume } C_x = 0.15$$

$$\frac{A_m}{A_{ts}} = 0.05$$

$$K_{ts} = \frac{C_x A_m}{A_{ts} \left(1 - \frac{A_m}{A_{ts}}\right)^3} = \underline{0.029}$$

C. Energy Ratio

Excluding the model resistance, $K_{ts} \text{ total} = 0.290$. $K_{ts} \text{ total}$ is increased by 10 percent to account for pressure losses due to screens in the settling chamber and leaks in the tunnel. The energy ratio is defined as the ratio of the flow power in the test section $\left(\frac{\rho V^3 A_{ts}}{2}\right)$ to the installed motor power (P), where V = flow speed in the test section. The installed motor power is equal to

$$\frac{\rho V^3 A_{ts} K_{ts}}{2\eta} \quad \text{where } K_{ts} \text{ is the total tunnel resistance coefficient and } \eta \text{ is the fan efficiency. Assuming } \eta = 90 \text{ percent and knowing } K_{ts} \text{ total} = (0.290)\left(\frac{11}{10}\right) = 0.319,$$

$$\text{Energy Ratio} = \left(\frac{\rho V^3 A_{ts}}{2}\right) / \left(\frac{\rho V^3 A_{ts} K_{ts}}{2\eta}\right) = \frac{\eta}{K_{ts}} = \frac{0.90}{0.319} = \underline{2.82}.$$

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